Description

Bearing for axially mounting a rotor of a gas turbine, and gas turbine

The invention relates to a bearing for axially mounting a rotor of a gas turbine, having a rotationally fixed bearing body which has a hydraulic piston arrangement for axially displacing the rotor from a first operating position into a second operating position, and having a hydraulic system fluidically connected to the hydraulic piston arrangement. The invention also relates to a gas turbine having such a bearing.

Bearings of the aforesaid type are known per se from the prior art. The bearing body of annular design preferably surrounding the rotor of a gas turbine serves for the arrangement of a plurality of hydraulic pistons. The latter are mounted against stop surfaces formed on the rotor, so that the rotor is supported in the axial direction.

Such a bearing for displacing the rotor of a gas turbine has been disclosed by US 2002/0009361. Once the gas turbine and its rotor have completely warmed up and thus the temperature-induced material expansions have stopped, the rotor is displaced by means of the bearing from a first operating position into a second operating position against the direction of flow of the hot working medium. As a result, in the turbine unit, the radial gaps formed between the moving blade tips and guide rings opposite the latter are minimized, so that a higher power output of the gas turbine is achieved and the losses via the moving blade tips are minimized.

The failure of hydraulic medium when the rotor is already arranged in the second operating position causes the rotor to be pushed back into the first operating position by the axial

thrust of the working medium, a factor which may lead to severe damage to the axial bearing, the rotor and the gas turbine.

The position of the rotor can be set by the position of the hydraulic pistons, which, depending on the set piston stroke, leads to an adjustment of the rotor of the gas turbine in the axial direction. The hydraulic piston arrangement therefore enables the rotor of the gas turbine to be oriented in relation to the bearing in accordance with the requirements and also enables it to be displaced from a first working position into a second working position. In a disadvantageous manner, however, the axial speed of the rotor occurring during the displacement of the rotor produces high dynamic forces, which may cause overloading of the bearing body, of the bearing housing and of the gas turbine. As a result of the axial displacement of the rotor, a total failure of the bearing may therefore occur. In this case, the moving of the rotor in the direction of flow is especially problematic, since very high thrusts act on the rotor in this displacement direction.

In addition, DE 23 57 881 A1 and US 4,915,510 have each disclosed an axial bearing which compensates for an unintentional axial displacement, caused by changing axial thrusts, of the rotor. Furthermore, DE 39 26 556 A1 shows a self-balancing axial bearing for asymmetrical axial movements of the rotor.

Based thereon, an object of the invention, while avoiding the aforesaid disadvantages, is to provide a bearing which absorbs the bearing forces occurring as a result of high dynamic thrusts of the rotor and ensures reliable mounting of the rotor. It is also an object of the invention to specify a gas turbine in this respect.

To achieve the first-mentioned object, the invention proposes a bearing of the type described above which is characterized in

that, to limit the displacement speed of the rotor, at least one restrictor for the hydraulic medium is provided between hydraulic piston arrangement and hydraulic system.

Owing to the fact that a restrictor is interposed according to the invention, the hydraulic medium displaced by the individual pistons is first of all directed through the restrictor, a factor which advantageously leads to a reduction in the kinetic energy and to a comparatively slow displacement of the rotor. The loads acting on the bearing body can thus be reduced, whereby the risk of overloading is minimized. Even at a maximum force acting on the rotor, kinetic energy can be sufficiently dissipated by the restrictor arranged between hydraulic piston arrangement and hydraulic system, so that overloading of the bearing as a result of dynamic forces of the rotor is prevented. Reliable mounting of the rotor of the gas turbine is thus ensured even during any possible occurrence of high dynamic thrusts.

The restrictor is advantageously arranged in the bearing body. Even in the unusual event of a line fracture in the hydraulic system, the hydraulic medium can only flow off quickly to a limited extent, which results in a low and thus non-damaging displacement speed of the rotor. The bearing, rotor and gas turbine are thus protected against defects which would be caused by an excessive displacement speed of the rotor. In this case, the restrictors are formed by flow constrictions.

Furthermore, in an advantageous development, the bearing can have at least one flow-control valve, designed as a restrictor, between hydraulic piston arrangement and hydraulic system. This protection likewise increases the safety of the entire system and in addition makes it possible for the flow velocity of the hydraulic medium and thus the displacement speed of the rotor to be set.

According to a further feature of the invention, provision is made for the hydraulic piston arrangement to have a plurality of pistons arranged in corresponding respective piston chambers. The arrangement of a plurality of hydraulic pistons advantageously achieves a more uniform introduction of force, which permits rectilinear and positionally accurate moving of the rotor of the gas turbine.

Furthermore, provision may be made with the invention for the piston chambers to be bores of cylindrical design. Of course, other geometrical configurations are also conceivable, but it has been found that in particular piston chambers of cylindrical design permit an optimized force distribution.

According to a special advantage of the invention, the piston chambers are fluidically connected to one another. This advantageously achieves a pressure balance between the individual piston chambers, thereby achieving, firstly, a uniform load distribution, but also, secondly, a uniformly rectilinear displacement of the rotor of the gas turbine. The individual piston chambers may in this case be fluidically connected to a common pressure space via appropriately designed bores or else may be fluidically connected to one another directly via appropriately designed bores. It is crucial that a pressure balance can be effected between the individual piston chambers.

According to a further feature of the invention, the hydraulic piston arrangement is of annular design and surrounds the rotor, of circular design in cross section, of the gas turbine. For optimized transmission of force from the hydraulic pistons to the rotor, the pistons are arranged equidistantly from one another with respect to the hydraulic piston arrangement of annular design, so that a uniform force distribution can be achieved. Depending on the configuration and size of the hydraulic piston arrangement of annular design, the common

pressure space connecting the piston chambers to one another may likewise be of annular design and may surround the hydraulic piston arrangement.

According to a further feature of the invention, two hydraulic piston arrangements formed separately from one another are provided and are arranged opposite one another on the bearing body. In this configuration of the bearing according to the invention, the bearing body has a total of two hydraulic piston arrangements, which, depending on the configuration, have in each case a plurality of pistons. The pistons of the first hydraulic piston arrangement interact with a first stop surface and the pistons of the second hydraulic piston arrangement interact with a second stop surface. During a displacement of the rotor of the gas turbine, the pistons of the one hydraulic piston arrangement are extended as a result of this arrangement described above, whereas the pistons of the other hydraulic piston arrangement are retracted. During a displacement of the rotor in the opposite direction, a piston displacement of the hydraulic piston arrangement is likewise effected in the opposite direction. With respect to the thrust direction of the rotor, the one hydraulic piston arrangement is designated as main track bearing and the other hydraulic piston arrangement is designated as secondary track bearing.

According to a further feature of the invention, the two hydraulic piston arrangements, that is to say the main track bearing and the secondary track bearing, are fluidically connected to one another. The hydraulic medium displaced from one of the two hydraulic piston arrangements as a result of a displacement of the rotor can thus be used for the pressure buildup inside the other hydraulic piston arrangement.

If the rotor of the gas turbine is displaced against the thrust direction, a controlled shutdown of the gas turbine can no longer be ensured in the event of a failure of the hydraulic

medium supply, for example due to a line fracture or the like. This is due to the fact that the hydraulic piston arrangement of the secondary track side of the bearing body, that is to say the secondary track mounting, can no longer be supplied with hydraulic medium. In order to be able to permit a specific shutdown of the gas turbine even in the event of a failure of the hydraulic medium supply, it is proposed according to a further feature of the invention that the two hydraulic piston arrangements be fluidically connected to one another with a 4/2-way directional control valve interposed. The arrangement of such a directional control valve advantageously makes it possible for hydraulic medium to be delivered from the cylinder chambers of the main track side of the bearing body into the cylinder chambers of the secondary track bearing even in the event of a failure of the hydraulic medium supply. For this purpose, the 4/2-way directional control valve is merely to be switched into a de-energized position. As a result of the thrust of the rotor of the gas turbine on the pistons of the main track bearing, the hydraulic medium is then delivered from the piston chambers of the hydraulic piston arrangement of the main track side into the piston chambers of the secondary track bearing. A controlled shutdown of the gas turbine can thus be ensured even in the event of a failure of the hydraulic medium supply.

In an especially advantageous manner, a gas turbine has a bearing having the aforesaid features.

Further advantages and features of the invention follow from the description with reference to figure 1, which shows in a partly sectioned side view a bearing designed according to the invention.

The bearing 1 according to the invention is shown in figure 1 in a partly sectioned side view. This bearing 1 serves to axially mount and displace a rotor 8 of a gas turbine and

comprises a bearing body 2 which serves to accommodate a first hydraulic piston arrangement 3 and a second hydraulic piston arrangement 4. With respect to the thrust direction 7 of the rotor 8, the hydraulic piston arrangement 3 is in this case arranged on the main track side 5 and the hydraulic piston arrangement 4 is arranged on the secondary track side 6 of the rotor 8.

Both the hydraulic piston arrangement 3 and the hydraulic piston arrangement 4 are formed by a plurality of pistons 23 guided in respective piston chambers 22. Via displaceably arranged intermediate elements, the pistons 23 of the hydraulic piston arrangement 3 act on the stop surface 24 formed on the rotor 8, and the pistons 23 of the hydraulic piston arrangement 4 act on the stop surface 25 likewise formed on the rotor 8, so that an axially displaceable mounting of the rotor 8 overall is formed. Via the arrangement of the hydraulic piston arrangements 3 and 4 in the bearing body 2, the rotor 8, for example of a gas turbine, can be positioned in a displaceable manner in the axial direction. For this purpose, the piston chambers 22 can be selectively filled with hydraulic medium, for example hydraulic oil, the hydraulic piston arrangement 3 and the hydraulic piston arrangement 4 being connected to a common hydraulic system 9 via the lines 10 and 11.

Components of the hydraulic system 9 are a tank 12, an accumulator 13, a hydraulic pump 14, check valves 15, 16 and 17, a 2/2-way directional control valve 18, a 4/2-way directional control valve 19 and adjustable flow-control valves 20 and 21. Restrictors 26 and 27 are provided between the hydraulic piston arrangements 3 and 4 and the hydraulic system 9. The restrictors 26, 27 are formed by flow constrictions, arranged directly in the bearing body 2, for the hydraulic medium without a line for hydraulic medium being interposed. In addition, the flow-control valves 20, 21 serve as further

restrictors between the hydraulic piston arrangement 3 or 4, respectively, and the hydraulic system 9.

The movement of the rotor 8 is achieved by its axial displacement relative to the bearing body 2 by hydraulic medium being pumped either into the hydraulic piston arrangement 3 or into the hydraulic piston arrangement 4. In the event of hydraulic medium being forced into the hydraulic piston arrangement 3, the pistons 23 extend in accordance with the filled quantity of hydraulic medium and act on the stop surface 24 of the rotor 8 via the elements arranged in between. This results in a displacement of the rotor 8 against the thrust direction 7, that is to say from a first operating position into a second operating position for reducing the radial gaps known in the prior art, thus to the left with respect to the image plane. Due to this longitudinal displacement of the rotor 8, the stop surface 25 acts on the hydraulic piston arrangement 4, and consequently hydraulic fluid is displaced from the corresponding piston chambers 22 and is fed via the line 11 to the hydraulic system 9.

According to the invention, in order to avoid a situation in which the axial speed of the rotor 8 occurring during the displacement of the rotor 8 produces excessive dynamic forces, which could cause overloading of the bearing body 2, the flow-control valves 20, 21 and restrictors 26, 27 embedded in the bearing body 2 are interposed between hydraulic piston arrangement 3, on the one hand, and hydraulic piston arrangement 4, on the other hand, and the hydraulic system 9. Even at a maximum force during operation, said flow-control valves 20, 21 and restrictors 26, 27 can sufficiently dissipate kinetic energy to the rotor, so that overloading of the bearing 1 as a result of the dynamic forces of the rotor 8 is prevented.

During the loading which occurs during the faultless operation of the gas turbine with an intended displacement of the rotor 8, the adjustable flow-control valves 20, 21 limit the flow velocity of the hydraulic medium to a predetermined value. The displacement of the rotor 8 both in the thrust direction of the hot gases and against the thrust direction is thus effected at the predetermined comparatively slow speed.

In the event of a fault in the hydraulic system 9, in the event of a failure of the flow-control valves 20, 21 or even in the event of a line fracture of the line 10, 11, connected to the bearing body 2, of the hydraulic system 9, the restrictors 26, 27 provided in the bearing body 2 limit the flow velocity of the hydraulic medium. The unforeseen displacement, taking place in the thrust direction 7, of the rotor 8 from the second operating position back into the first operating position is then effected at a speed which protects the bearing body 2 from damage and which may be greater than the speed which is desired during faultless operation and which is set by means of the flow-control valves.

The flow constrictions in the bearing body, which are formed by the restrictors 26, 27, are calculated for an assumed maximum load which is higher than the operating load.

The restrictors 26, 27 limit the displacement speed of the rotor only in the event of a fault, whereas the flow-control valves 20, 21 limit the admissible displacement speed of the rotor during an intended displacement of the latter.

Furthermore, provision is made according to the invention for the hydraulic piston arrangement 3 and the hydraulic piston arrangement 4 to be fluidically connected to one another via the hydraulic system 9 with the 4/2-way directional control valve 19 arranged in between.

In addition, in bearing units previously known from the prior art, it is not possible in the event of the failure of the hydraulic medium supply in the case of a rotor displaced against the thrust direction to ensure a controlled shutdown of the gas turbine, since the hydraulic piston arrangement 4 on the secondary track side 6 of the bearing 1 cannot be supplied with hydraulic medium. This is remedied here by the 4/2-way directional control valve arranged according to the invention in between main track side 5 and secondary track side 6. This is because this 4/2-way directional control valve can be switched into a de-energized position in the event of the failure of the hydraulic medium supply. To be precise, as a result of the thrust of the rotor 8 on the pistons 23 of the hydraulic piston arrangement 3, hydraulic medium is delivered from the piston chambers 22 of the main track side 5 via the hydraulic system 9 into the piston chambers 22 of the secondary track side 6. The pressure on the side of the secondary track therefore builds up, so that, in the event of a pressure drop, the rotor comes to a stop where it would likewise be if the bearing were not hydraulically adjustable. As a result, an emergency-running displacement of the rotor 8 can be achieved even if the hydraulic medium supply is interrupted, and consequently a controlled shutdown of the gas turbine remains possible.